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(54) **ENGINE LUBRICATION CONTROL SYSTEM**

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F01M 1/16 (2006.01)

(52) **U.S. Cl.**
CPC ***F01M 1/16*** (2013.01)

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USPC .... 123/196 R, 196 CP, 196 V, 196 AA; 137/565.16, 625.65, 625.25, 528
See application file for complete search history.

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(57) **ABSTRACT**

The present invention provides an engine lubrication control system including an engine, an oil pump which is driven by the engine, an oil circuit which extends downstream from the oil pump, and a plurality of oil branch supply paths which branch from the oil circuit and supply oil to each part of the engine. An electronically-controlled first hydraulic control valve which controls, in a stepwise manner, a discharge pressure of the oil pump relative to a speed of the engine is disposed on the oil circuit, a hydraulically-driven second hydraulic control valve is disposed on at least one of the plurality of oil branch supply paths, and a downstream hydraulic pressure of the second hydraulic control valve is controlled to be lower than a downstream hydraulic pressure of the first hydraulic control valve of the oil circuit at least across a predetermined engine speed range.

18 Claims, 8 Drawing Sheets

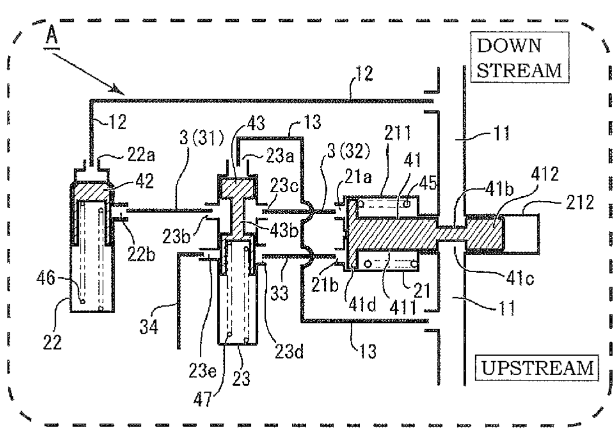
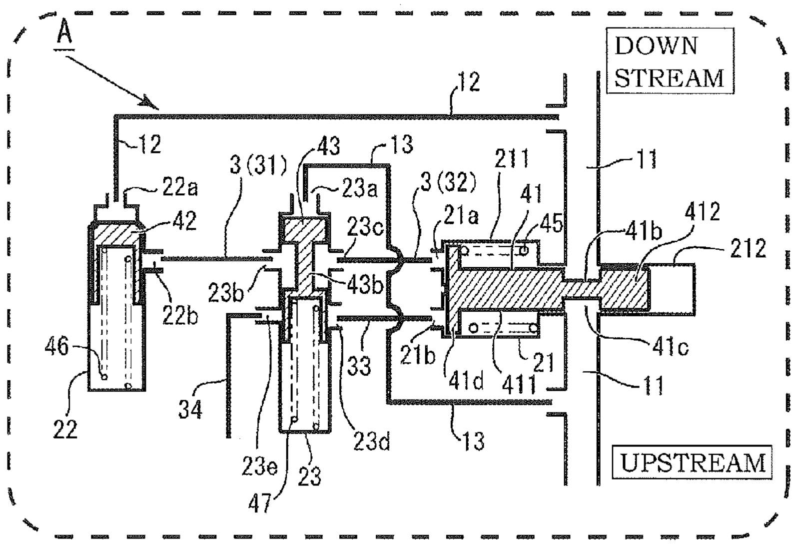


Fig. 1B



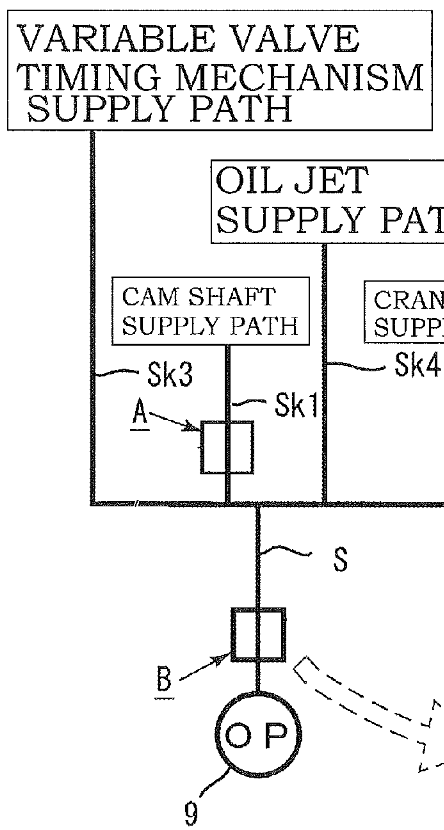
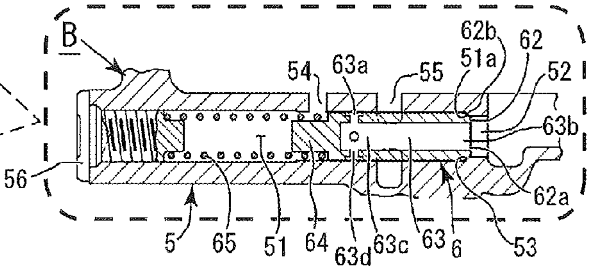


Fig. 1A

Fig. 1C



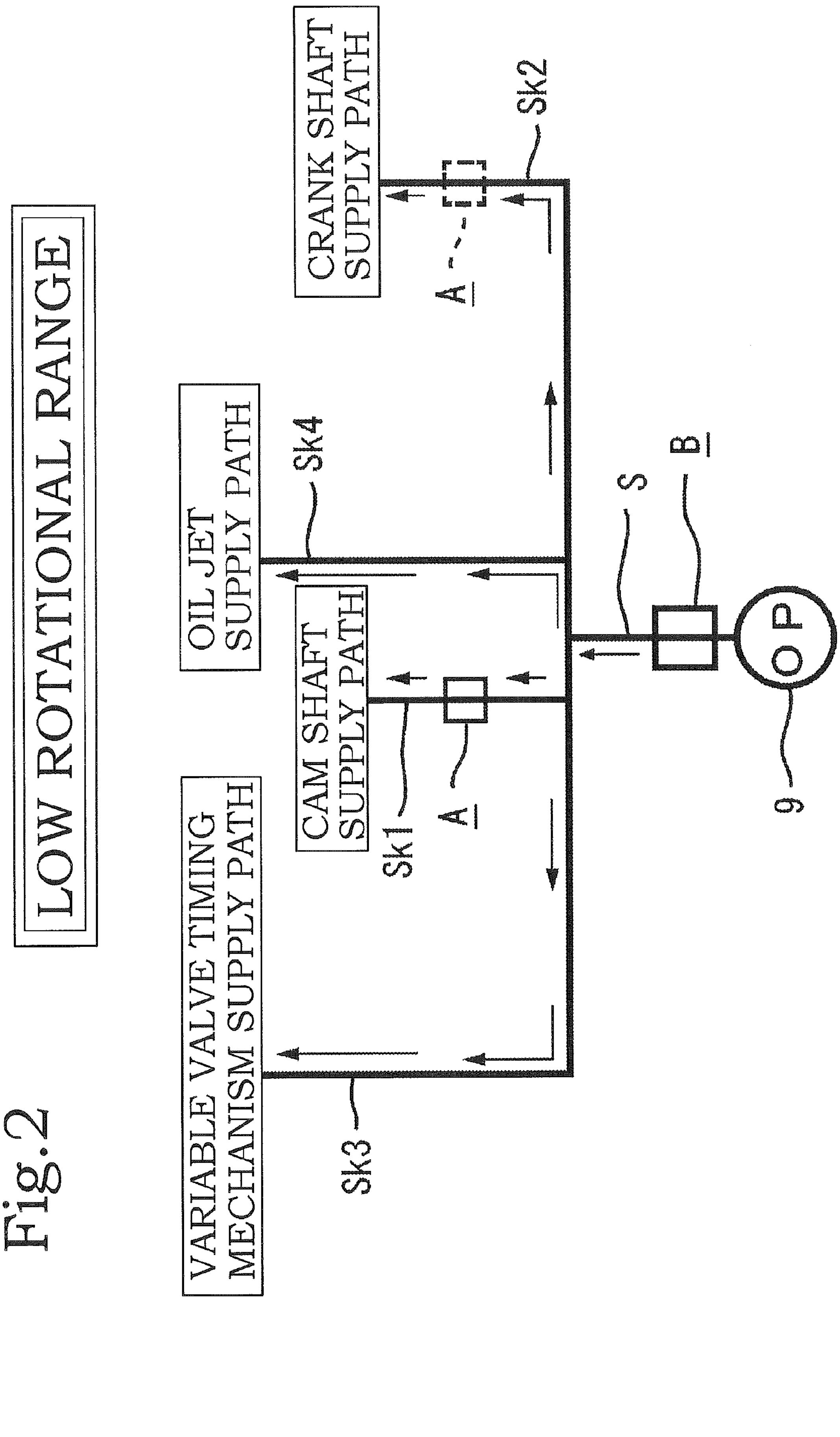
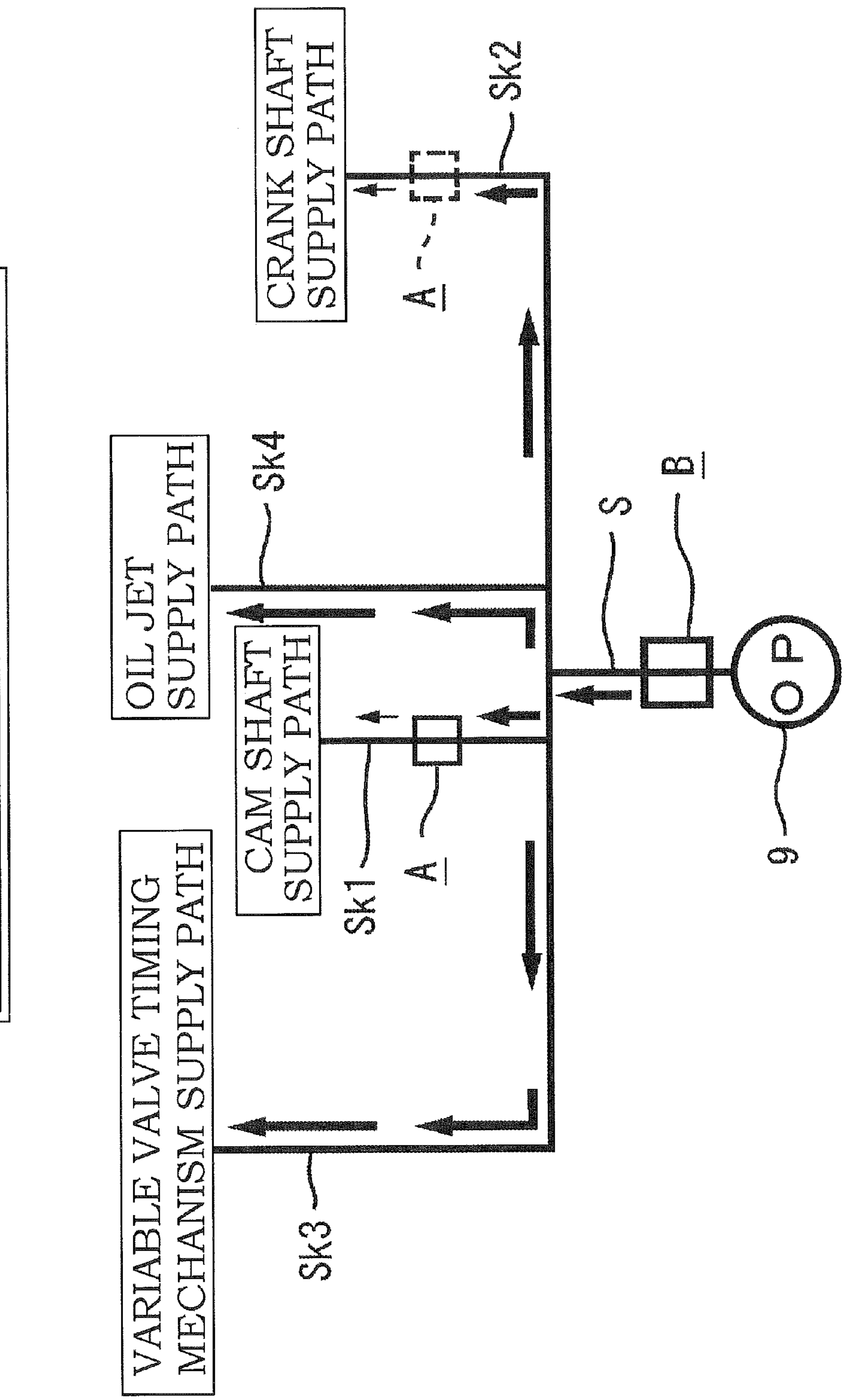

LOW ROTATIONAL RANGE
CRANK SHAFT SUPPLY PATH
Sk2
A
OIL JET SUPPLY PATH
Sk4
S
B
CAM SHAFT SUPPLY PATH
OP
A
Sk1
9
VARIABLE VALVE TIMING MECHANISM SUPPLY PATH
Sk3

Fig.2

Fig.3


MID ROTATIONAL RANGE
VARIABLE VALVE TIMING
MECHANISM SUPPLY PATH
CAM SHAFT
SUPPLY PATH
OIL JET
SUPPLY PATH
CRANK SHAFT
SUPPLY PATH
Sk1
Sk2
Sk3
Sk4
A
A
B
S
OP
9

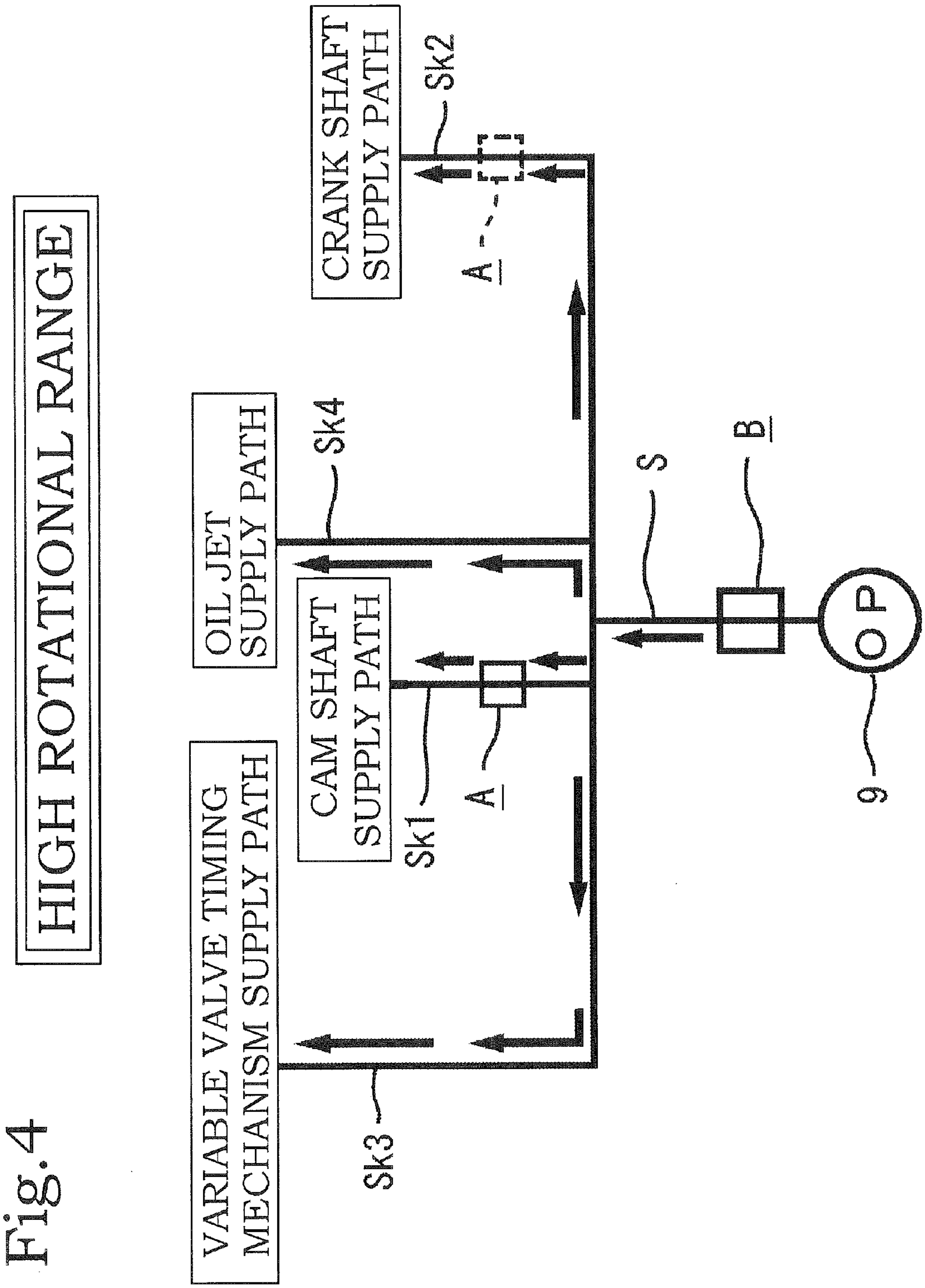
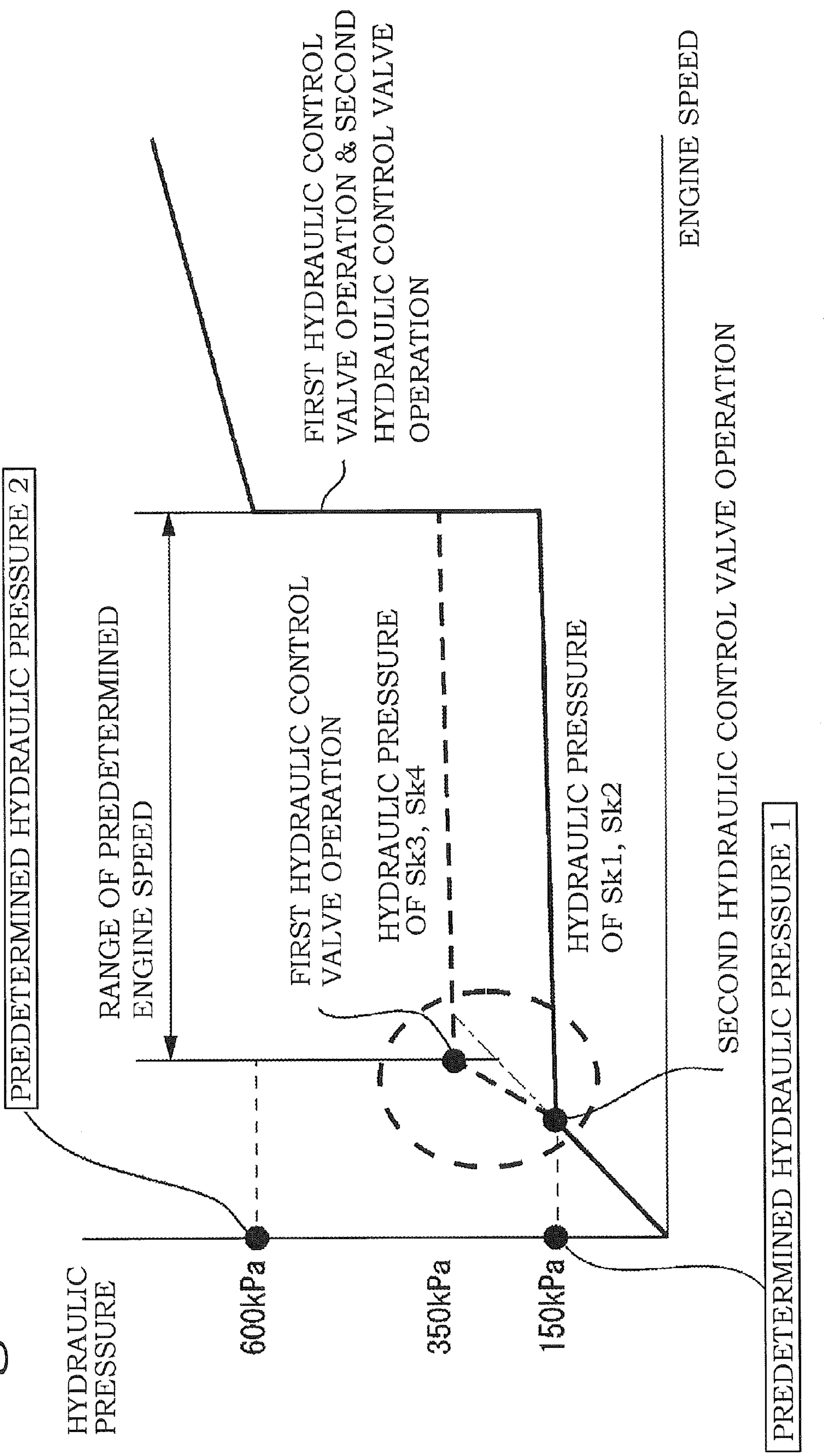

HIGH ROTATIONAL RANGE
VARIABLE VALVE TIMING MECHANISM SUPPLY PATH
CAM SHAFT SUPPLY PATH
OIL JET SUPPLY PATH
CRANK SHAFT SUPPLY PATH
Sk1
Sk2
Sk3
Sk4
A
A
B
S
OP
9

Fig. 4

Fig. 5



HYDRAULIC PRESSURE
600kPa
350kPa
150kPa
PREDETERMINED HYDRAULIC PRESSURE 2
RANGE OF PREDETERMINED ENGINE SPEED
FIRST HYDRAULIC CONTROL VALVE OPERATION
HYDRAULIC PRESSURE OF Sk3, Sk4
HYDRAULIC PRESSURE OF Sk1, Sk2
FIRST HYDRAULIC CONTROL VALVE OPERATION & SECOND HYDRAULIC CONTROL VALVE OPERATION
SECOND HYDRAULIC CONTROL VALVE OPERATION
PREDETERMINED HYDRAULIC PRESSURE 1
ENGINE SPEED

Fig.6A

LOW ROTATIONAL RANGE

FIRST HYDRAULIC CONTROL VALVE

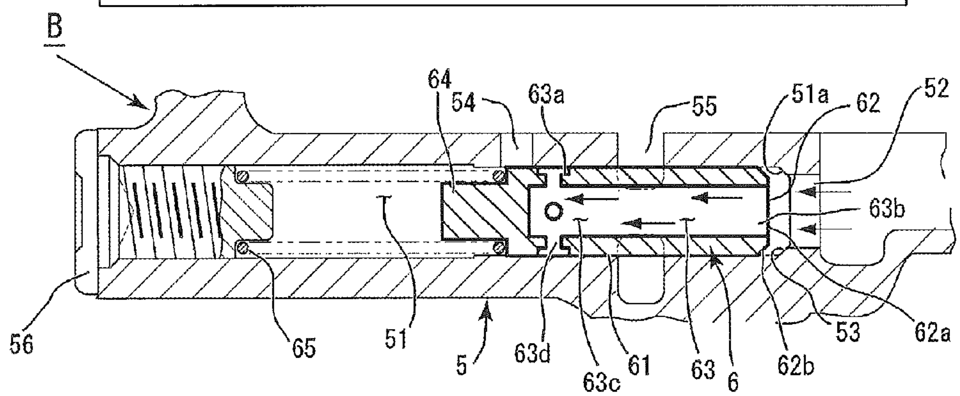


Fig.6B

SECOND HYDRAULIC CONTROL VALVE

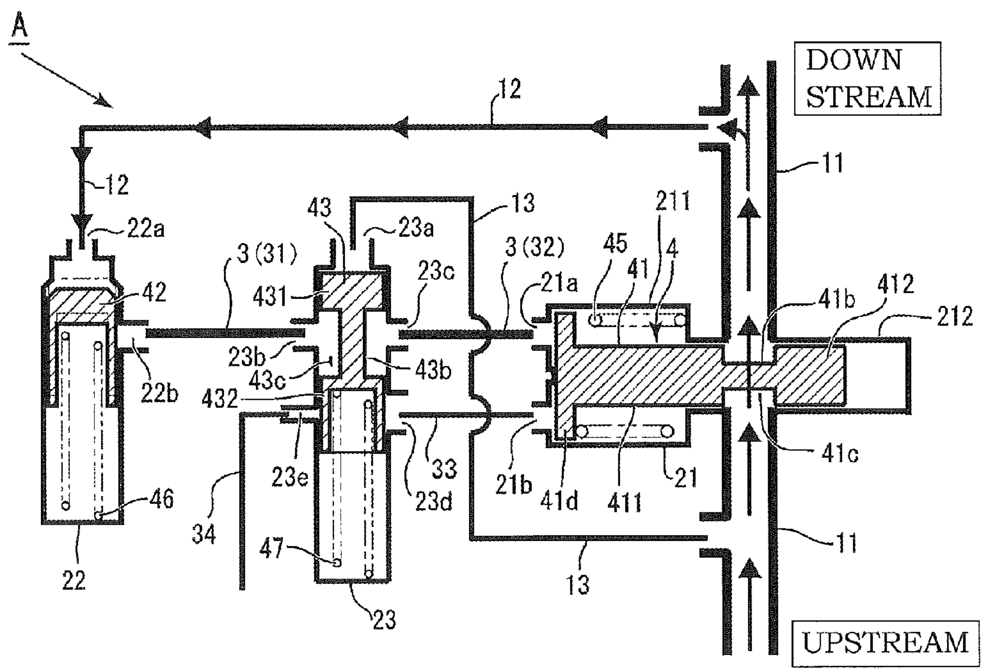


Fig.7A

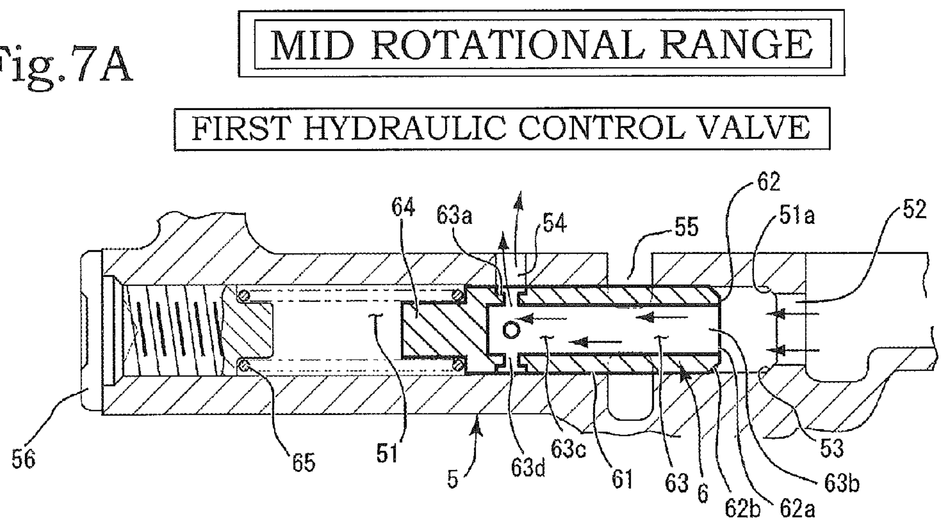


Fig.7B

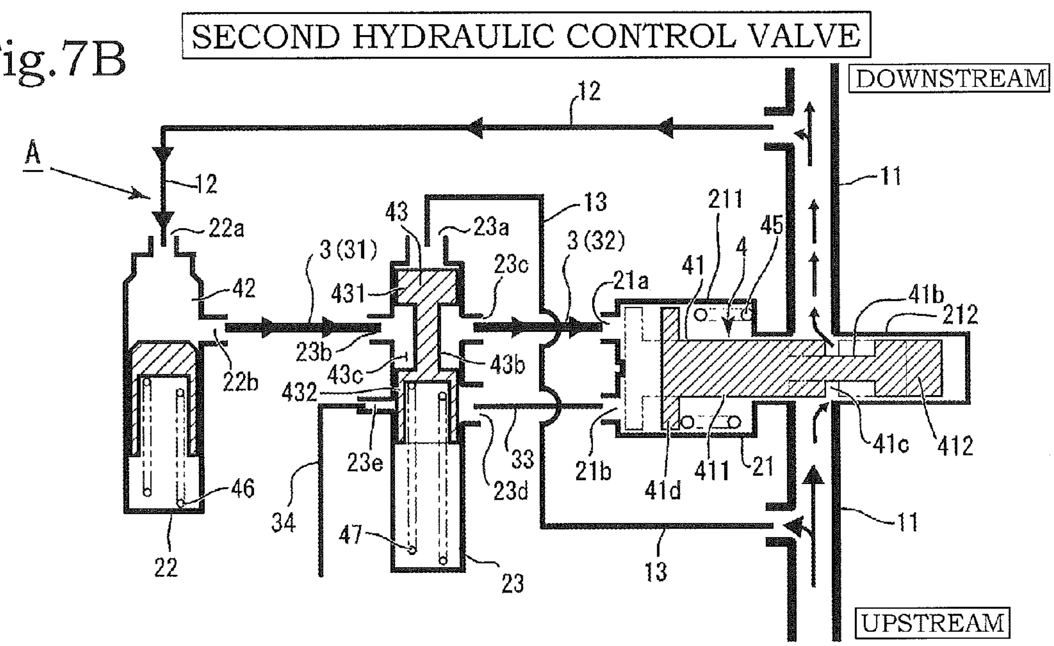


Fig.7C

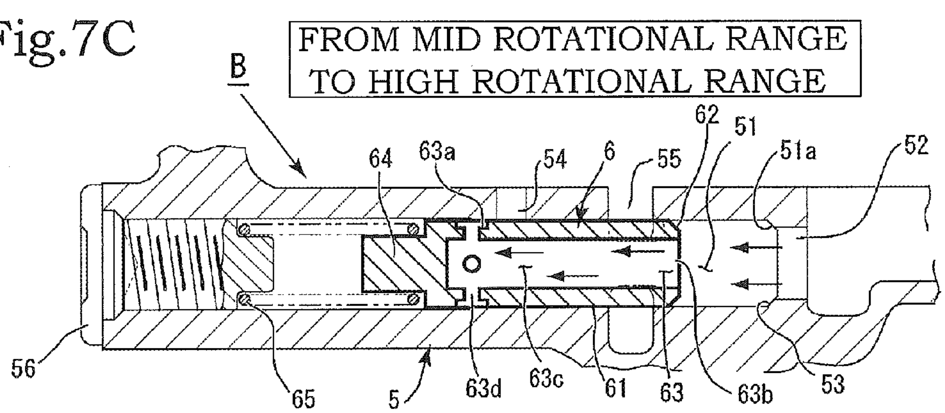


Fig.8A

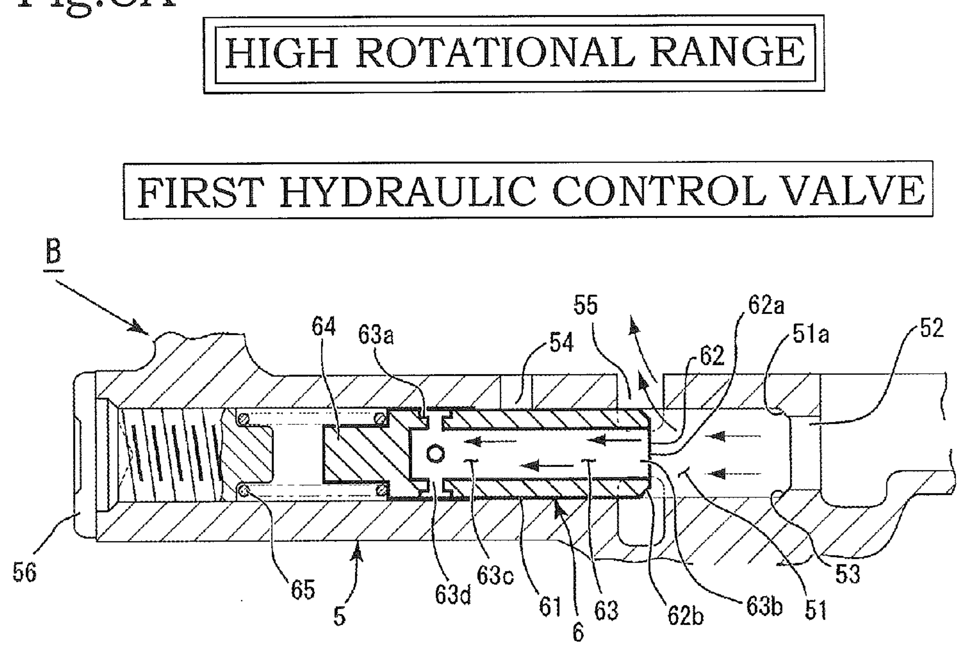
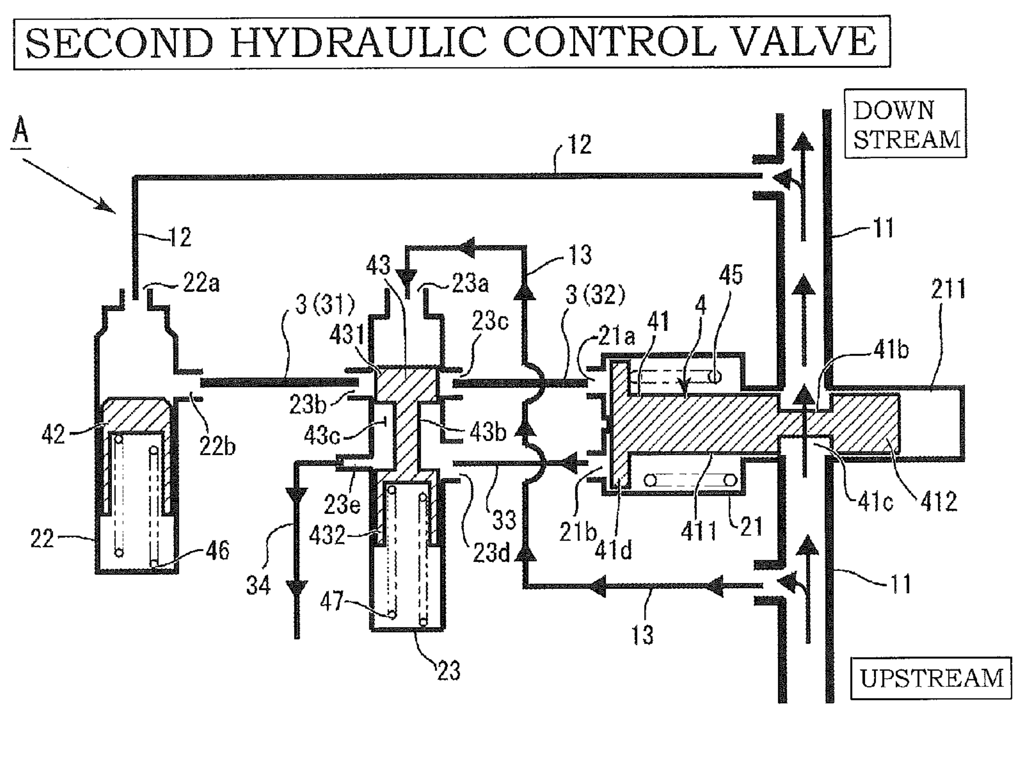

HIGH ROTATIONAL RANGE
FIRST HYDRAULIC CONTROL VALVE
B
64
63a
54
55
62a
62
51a
52
56
65
5
63d
63c
61
6
63
62b
63b
51
53

Fig.8B


SECOND HYDRAULIC CONTROL VALVE
A
12
DOWN STREAM
11
13
43
23a
22a
3 (31)
431
23c
3 (32)
21a
41
4
45
211
41b
23b
43c
43b
42
22b
23e
33
21b
41d
411
21
41c
412
22
46
432
23d
34
47
23
UPSTREAM

ENGINE LUBRICATION CONTROL SYSTEM

BACKGROUND OF THE INVENTION

1. Field of the Invention

The present invention relates to an engine lubrication control system for adjusting the hydraulic pressure that is supplied to respective channels in a lubricating oil feeding device of an engine, or more particularly in a lubricating oil feeding device provided with a cam shaft supply channel for feeding lubricating oil to a cam journal or the like of a cylinder head, and a crank shaft supply channel for feeding lubricating oil to a crank shaft, a connecting rod or the like of a cylinder block.

2. Description of the Related Art

Conventionally, since oil that is needed by sliding parts of the engine such as a crank shaft and a cam shaft or cam shaft mechanical sections is supplied by an oil pump that is driven by the engine, the pressure of the oil that is supplied from the oil pump to the respective components of the engine will change substantially in proportion to the speed of the engine. Thus, depending on the engine speed, there are cases where the discharge pressure becomes greater than necessary, and there is a problem in that the friction of the oil pump increases more than necessary and unneeded work is thereby increased. In view of this, attempts are being made to achieve an appropriate discharge pressure in accordance with the engine speed.

As a lubrication control system for achieving the foregoing object, there is the type disclosed in, for example, Japanese Patent Application Publication No. 2009-264241. Japanese Patent Application Publication No. 2009-264241 is now briefly explained. The reference numerals used in the explanation are cited as is from Japanese Patent Application Publication No. 2009-264241. Foremost, oil is pumped up from an oil pan **10** by an oil pump **12**, and fed to a first oil supply route **16***a* (lower route), and a second oil supply route **16***b* (upper route).

The first oil supply route **16***a* is mainly a route for supplying oil to a bearing **18** of the crank shaft, and the second oil supply route **16***b* is a route for supplying oil, for instance, to a valve gear **20**. A hydraulic pressure control valve **22** for controlling the oil content to be supplied to the bearing **18** of the crank shaft is disposed above the first oil supply route **16***a*. The hydraulic pressure control valve **22** is configured so that its output hydraulic pressure is controlled by the control unit **24**.

The control unit **24** is controlled by an engine speed sensor **26**, an engine load sensor **28**, an oil temperature sensor **30**, and a hydraulic pressure sensor **32**. Provided is a relief valve **34** which relieves the excessive hydraulic pressure from the oil route between the oil pump **12** and a filter **14** to the oil pan **10** when the hydraulic pressure exceeds a predetermined value. In the foregoing configuration, the hydraulic pressure control valve **22** is electronically controlled by the control unit **24**.

SUMMARY OF THE INVENTION

In Japanese Patent Application Publication No. 2009-264241 and conventional technology comprising a similar configuration, the hydraulic pressure that is supplied to the cam shaft is controlled by the relief valve to be a substantially constant hydraulic pressure at a predetermined engine speed or higher. However, with this kind of configuration, the hydraulic pressure that is controlled by the relief valve needs

to be a high pressure during the high rotation and high load of the engine so that the lubrication of the cam shaft will remain sufficient.

Thus, the hydraulic pressure that is supplied to the camshaft in a mid rotational range of the engine becomes the hydraulic pressure corresponding to the engine speed. Nevertheless, since the hydraulic pressure that is required in the cam shaft in a mid rotational range of the engine is generally lower than the hydraulic pressure corresponding to the engine speed, the oil pump will supply greater hydraulic pressure than necessary, and there is a problem in that it is not possible to reduce the friction of the oil pump.

Thus, as a result of intense study to overcome the foregoing problem, the present inventors discovered that it is possible to resolve the foregoing problem by causing the first aspect of the present invention to be an engine lubrication control system including an engine, an oil pump which is driven by the engine, an oil circuit which extends downstream from the oil pump, and a plurality of oil branch supply paths which branch from the oil circuit and supply oil to each part of the engine, wherein a hydraulically-driven first hydraulic control valve which controls, in a stepwise manner, a discharge pressure of the oil pump relative to a speed of the engine is disposed on the oil circuit, a hydraulically-driven second hydraulic control valve is disposed on at least one of the plurality of oil branch supply paths, and a downstream hydraulic pressure of the second hydraulic control valve is controlled to be lower than a downstream hydraulic pressure of the first hydraulic control valve of the oil circuit at least across a range of a predetermined engine speed.

The foregoing problem was additionally resolved by causing the second aspect of the present invention to be, in the first aspect, an engine lubrication control system in which the second hydraulic control valve is disposed on a crank shaft supply path or a cam shaft supply path among the plurality of oil branch supply paths. The foregoing problem was additionally resolved by causing the third aspect of the present invention to be, in the first aspect or the second aspect, an engine lubrication control system in which the downstream hydraulic pressure of the first hydraulic control valve of the oil circuit is controlled to be substantially the same as the downstream hydraulic pressure of the second hydraulic control valve, at an engine speed that is higher than the predetermined engine speed range.

The foregoing problem was additionally resolved by causing the fourth aspect of the present invention to be, in one aspect among the first to third aspects, an engine lubrication control system in which the engine speed when an operation the second hydraulic control valve is started is lower than the engine speed when an operation of the first hydraulic control valve is started.

The foregoing problem was additionally resolved by causing the fifth aspect of the present invention to be, in one aspect among the first to fourth aspects, an engine lubrication control system in which the second hydraulic control valve includes a channel cross-sectional area adjustment spool which changes a channel cross-sectional area of a main channel of the crank shaft supply path, and the channel cross-sectional area adjustment spool decreases the channel cross-sectional area of the main channel when a downstream hydraulic pressure of the channel cross-sectional area adjustment spool is greater than a predetermined hydraulic pressure value **1**, and the channel cross-sectional area adjustment spool is restored such that the channel cross-sectional area of the main channel is maximized when a hydraulic pressure that is more upstream than the channel cross-sectional area adjustment spool is a predetermined hydraulic pressure value **2**, which is greater than the predetermined hydraulic pressure value **1**.

According to the first aspect of the present invention, in a predetermined engine speed range; for instance, in a mid rotational range, the hydraulic pressure that is supplied to the respective parts of the engine is controlled, by the first hydraulic control valve, to be lower than the discharge pressure of the oil pump which is substantially proportionate to the engine speed. Moreover, while the hydraulic pressure that is needed in the respective parts of the engine differs for each part, the second hydraulic control valve disposed on the oil branch supply path can further decrease the hydraulic pressure of parts in which their functions can be satisfied even with a low hydraulic pressure.

Consequently, in a predetermined engine speed range, by not disposing the second hydraulic control valve to portions that require a relatively high hydraulic pressure, and disposing the second hydraulic control valve to portions in which their functions can be satisfied even with a low hydraulic pressure to achieve a low hydraulic pressure, an appropriate hydraulic pressure can be distributed to the respective parts of the engine.

Moreover, since the minimum required hydraulic pressure can be supplied to the respective parts of the engine, the work of the oil pump can be minimized, and this will contribute to the improvement in efficiency. In addition, since the hydraulically-driven second hydraulic control valve is driven in conjunction with the change in the hydraulic pressure of the electronically-controlled first hydraulic control valve capable of performing accurate control, accurate control can also be performed by the hydraulically-driven second hydraulic control valve which is easily influenced by disturbance such as the oil temperature.

The second aspect of the present invention yields substantially the same effect as first aspect. Moreover, since bearings of the crank shaft and cam shaft or the like are subject to considerably reduced sliding resistance based on the decreased hydraulic pressure due to the operation of the second hydraulic control valve, fuel efficiency can be improved.

According to the third aspect of the present invention, by raising the downstream hydraulic pressure of the second hydraulic control valve, which is of a low hydraulic pressure, to be substantially the same as the downstream hydraulic pressure of the first hydraulic control valve, sufficient lubrication and cooling can be performed even when the engine is in a state of high rotation and high load.

According to the fourth aspect of the present invention, by causing the second hydraulic control valve to start its operation at a lower engine speed, the oil groove of the oil branch supply path on which the second hydraulic control valve is disposed becomes constricted. Consequently, since more oil will flow to other oil branch supply paths, the hydraulic pressure of the oil flowing through the other oil branch supply paths will increase.

If a variable valve timing mechanism or a device which operates at a predetermined hydraulic pressure of an oil jet or the like is disposed on the other oil branch supply paths, the hydraulic pressure required for that device can be secured from the low rotation side, and the range of the engine speed in which the device will operate can be expanded.

According to the fifth aspect of the present invention, since the second hydraulic control valve contracts and restores (expands) the channel cross-sectional area of the main channel by directly using the upstream and downstream hydraulic pressures of the channel cross-sectional area adjustment spool, the operation of the channel cross-sectional area adjustment spool becomes accurate and highly responsive, and the friction of the oil pump can be decreased without impairing the lubrication of the crank shaft.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1**A is a configuration diagram of the engine lubrication control system of the present invention, FIG. **1**B is a schematic diagram of the configuration of the second hydraulic control valve of FIG. **1**A, and FIG. **1**C is a schematic diagram of the configuration of the first hydraulic control valve (electronically controlled 2-stage relief valve) of FIG. **1**A;

FIG. **2** is a schematic diagram showing the state of the oil in a low rotational range of the engine lubrication control system in the present invention;

FIG. **3** is a schematic diagram showing the state of the oil in a mid rotational range of the engine lubrication control system in the present invention;

FIG. **4** is a schematic diagram showing the state of the oil in a high rotational range of the engine lubrication control system in the present invention;

FIG. **5** is a graph showing the characteristics of the engine lubrication control system in the present invention;

FIG. **6**A is a schematic diagram showing the operating state of the first hydraulic control valve (2-stage relief valve) in a low rotational range, and FIG. **6**B is an operating state diagram of the second hydraulic control valve in a low rotational range;

FIG. **7**A is a schematic diagram showing the operating state of the first hydraulic control valve (2-stage relief valve) in a mid rotational range, FIG. **7**B is an operating state diagram of the second hydraulic control valve in a mid rotational range, and FIG. **7**C is a schematic diagram showing the operating state of the first hydraulic control valve (2-stage relief valve) from a mid rotational range to a high rotational range; and

FIG. **8**A is a schematic diagram showing the operating state of the first hydraulic control valve (2-stage relief valve) in a high rotational range, and FIG. **8**B is an operating state diagram of the second hydraulic control valve in a high rotational range.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Embodiments of the present invention are now explained with reference to the drawings. In the control system of the present invention, the circuit through which oil flows is configured from one oil circuit S, and a plurality of oil branch supply paths Sk (refer to FIG. **1**A, FIG. **2** to FIG. **4**). The oil circuit S is positioned upstream, and the oil branch supply paths Sk are positioned downstream. The circuit includes a plurality of oil branch supply paths Sk which branch from the oil circuit S and supply oil to each part of the engine.

In addition, the plurality of oil branch supply paths Sk specifically include a cam shaft supply path Sk**1** and a crank shaft supply path Sk**2** which supply oil on the downstream side of the oil pump **9**, and are also sometimes provided with a variable valve timing mechanism supply path Sk**3**, or an oil jet supply path Sk**4** which sprays oil to the lower face of the piston of the engine.

In the oil branch supply path Sk, the crank shaft supply path Sk**2** is mainly used for feeding oil to the bearings of the crank shaft or the like in a lower area of the engine, and the cam shaft supply path Sk**1** is a path for feeding oil to the valve gear of the engine and the like.

The oil circuit S is provided with a first hydraulic control valve B. Moreover, at least one of the plurality of oil branch supply paths Sk is provided with a second hydraulic control valve A. In other words, the second hydraulic control valve A is provided to several or all of the plurality of oil branch supply paths Sk.

The second hydraulic control valve A controls the hydraulic pressure of the oil branch supply path Sk to be lower than the hydraulic pressure controlled by the first hydraulic control valve B across a predetermined engine speed range. A configuration where the second hydraulic control valve A is provided only to the cam shaft supply path Sk**1** and the crank shaft supply path Sk**2** of the oil branch supply path Sk is explained below.

In the present invention, the oil pump **9** is a mechanically-driven oil pump **9**. Note that the illustration of the engine is omitted. As a specific example of the second hydraulic control valve A, the second hydraulic control valve A is provided to the crank shaft supply path S**2** on the oil branch supply path Sk, and the first hydraulic control valve (2-stage relief valve) B is provided to the cam shaft supply path S**1**. In addition, the second hydraulic control valve A is disposed more downstream than the first hydraulic control valve B (2-stage relief valve) with the position of the oil pump **9** as the reference.

The second hydraulic control valve A is configured from a housing not shown, a channel cross-sectional area adjustment spool **41**, a channel on/off valve **42**, a channel on/off spool **43**, and elastic members **45**, **46**, **47** that elastically bias the foregoing valves. A main channel **11** is formed in the housing. The main channel **11** configures a part of the oil branch supply paths Sk.

Formed in the housing are a channel cross-sectional area adjustment spool chamber **21**, a channel on/off valve chamber **22** and a channel on/off spool chamber **23**. The channel cross-sectional area adjustment spool chamber **21** is formed at substantially the center portion of the main channel **11**, and more specifically is a room that is formed to intersect, in an orthogonal state, the middle portion of the main channel **11**, and is separated into two rooms by the main channel **11**. Mounted on the channel cross-sectional area adjustment spool chamber **21** is the channel cross-sectional area adjustment spool **41** described later.

Moreover, a downstream branch channel **12** is formed at a location that is positioned more downstream than the position of the channel cross-sectional area adjustment spool chamber **21** in the main channel **11**, and an upstream branch channel **13** is formed more upstream than the channel cross-sectional area adjustment spool chamber **21**.

The channel on/off valve chamber **22** is in communication with the downstream side of the main channel **11** via the downstream branch channel **12**. Moreover, the channel on/off spool chamber **23** is in communication with the upstream side of the main channel **11** via the upstream branch channel **13**. Specifically, the downstream branch channel **12** is in communication with an apex opening **22**_a_ of the channel on/off valve chamber **22** in the axial direction, and the upstream branch channel **13** is in communication with an apex opening **23**_a_ formed at the apex of the channel on/off spool chamber **23** in the axial direction.

A communication channel **3** is formed between the channel on/off valve chamber **22** and the channel cross-sectional area adjustment spool chamber **21**, and the channel on/off valve chamber **22** and the channel cross-sectional area adjustment spool chamber **21** are in communication via the communication channel **3**. The channel on/off spool chamber **23** is disposed at the middle portion of the communication channel **3**. That is, the communication channel **3** is configured to be separated into two by the channel on/off spool chamber **23**.

In addition, with the communication channel **3**, the channel between the channel on/off valve chamber **22** and the channel on/off spool chamber **23** is referred to as a first communication channel **31**, and the channel between the channel on/off spool chamber **23** and the channel cross-sectional area adjustment spool chamber **21** is referred to as a second communication channel **32**. One end of the first communication channel **31** is in communication with a lateral outlet **22**_b_ formed on a lateral face that is orthogonal to the channel on/off valve chamber **22** in the axial direction.

Moreover, the other end of the first communication channel **31** is in communication with a lateral inlet **23**_b_ formed on a lateral face that is orthogonal to the channel on/off spool chamber **23** in the axial direction. In addition, one end of the second communication channel **32** is in communication with a lateral outlet **23**_c_ formed on a lateral face that is orthogonal to the channel on/off spool chamber **23** in the axial direction. Moreover, the other end of the second communication channel **32** is in communication with an apex inlet **21**_a_ formed at the apex of the channel cross-sectional area adjustment spool chamber **21** in the axial direction.

In addition, a drain channel **33** is formed, in a communicating manner, between the channel on/off spool chamber **23** and the channel cross-sectional area adjustment spool chamber **21** at a position along the axial direction that is different from the second communication channel **32**. Specifically, an apex outlet **21**_b_ is formed at a position that is different from the apex inlet **21**_a_ at the apex of the channel cross-sectional area adjustment spool chamber **21**, a drain inlet **23**_d_ is formed at a position that is lower than the lateral outlet **23**_c_ in the axial direction one a lateral face that is orthogonal to the channel on/off spool chamber **23** in the axial direction, and the drain channel **33** is formed between the apex outlet **21**_b_ and the drain inlet **23**_d_.

Moreover, a drain outlet **23**_e_ is formed on the channel on/off spool chamber **23** at a position that is the same as the drain inlet **23**_d_ in the axial direction but different in the peripheral direction, and a discharge channel **34** which communicates with the outside of the housing is formed from the drain outlet **23**_e_.

Mounted on the channel cross-sectional area adjustment spool chamber **21** is the channel cross-sectional area adjustment spool **41**. The channel cross-sectional area adjustment spool **41** is mounted on the channel cross-sectional area adjustment spool chamber **21** slidably in the axial direction and so as to cut across the main channel **11** in a substantially orthogonal state. In addition, the channel cross-sectional area adjustment spool **41** functions to control the flow rate and pressure of the oil flowing in the main channel **11** by sliding in the axial direction and constricting the channel cross-sectional area of the main channel **11**.

The channel cross-sectional area adjustment spool **41** is configured from a first sliding part **411** that is inserted into the main chamber part **211**, a second sliding part **412** that is inserted into the sub chamber part **212**, a constricted part **41**_b_ that communicates the first sliding part **411** and the second sliding part **412**, and a large diameter flange-shaped part **41**_d_. The outer diameter of the first sliding part **411** and the second sliding part **412** is formed to be substantially equal to or slightly smaller than the inner diameter of the main channel **11**.

The constricted part **41**_b_ is formed to be smaller than the outer diameter of the first sliding part **411** and the second sliding part **412**. Moreover, the large diameter flange-shaped part **41**_d_ is formed at the end of the first sliding part **411** and formed to be larger than the outer diameter of the first sliding part **411**. The periphery of the constricted part **41***b* is an opening area **41***c*.

The channel cross-sectional area adjustment spool **41** is normally subject to the elastic biasing force of the elastic member **45** so that the constricted part **41***b* cuts across within the main channel **11** and the channel cross-sectional area of the main channel **11** is fully opened to maximum. As an embodiment of the elastic member **45**, a coil spring is mainly used. Moreover, a fully opened state of the main channel **11** refers to a state where only the constricted part **41***b* of the channel cross-sectional area adjustment spool **41** cuts across within the main channel **11**, and a state where the oil flows to the opening area **41***c*.

In addition, as a result of oil flowing from the apex inlet **21***a* of the channel cross-sectional area adjustment spool chamber **21**, the large diameter flange-shaped part **41***d* of the channel cross-sectional area adjustment spool **41** is pressed by the pressure from the oil flowing through the communication channel **3**, and the channel cross-sectional area adjustment spool **41** slides in the axial direction against the elastic biasing force of the elastic member **45**.

Consequently, the protrusion of the constricted part **41***b* will decrease while the protrusion of the first sliding part **411** will increase in the main channel **11**, the channel cross-sectional area of the main channel **11** is contracted from a fully open state, and the cross-sectional area of the main channel **11** is constructed and the flow rate and pressure of the oil will decrease (refer to FIG. **7**B). Moreover, the first sliding part **411** is used for contracting the channel cross-sectional area of the main channel **11**, and is not used for completely blocking the flow of oil, and reduces the flow rate and pressure of the oil.

Moreover, a channel on/off valve **42** is mounted on the channel on/off valve chamber **22**. The channel on/off valve **42** functions as an on/off valve for blocking and communicating the downstream branch channel **12** and the first communication channel **31** configuring the communication channel **3**. In addition, the channel on/off valve **42** is normally pressed toward the apex of the channel on/off valve chamber **22** in the axial direction by the elastic biasing force of the elastic member **46**, and is positioned at the apex of the channel on/off valve chamber **22**.

This state shall be the initial state of the channel on/off valve **42**. The channel on/off valve **42** is blocking the downstream branch channel **12** and the first communication channel **31** in a state of being positioned at the apex of the channel on/off valve chamber **22**; that is, in the initial state.

A channel on/off spool **43** is disposed on the channel on/off spool chamber **23**. The channel on/off spool **43** functions to communication and block the first communication channel **31** and the second communication channel **32** configuring the communication channel **3**. The channel on/off spool **43** is configured from a first sliding part **431**, a second sliding part **432** and a constricted part **43***b* that connects the first sliding part **431** and the second sliding part **432** and has a diameter that is smaller than the outer diameter of the first sliding part **431** and the second sliding part **432**. An opening area **43***c* is formed with the constricted part **43***b* and the inner wall of the channel on/off spool chamber **23**.

The channel on/off spool **43** is normally pressed toward the apex of the channel on/off spool chamber **23** by the elastic biasing force of the elastic member **47**, and is positioned at the apex of the channel on/off spool chamber **23**. This state shall be the initial state of the channel on/off spool **43**. The elastic member **46** and the elastic member **47** are mainly configured from coil springs.

The constricted part **43***b* is positioned at the lateral inlet **23***b* and the lateral outlet **23***c* when the channel on/off spool **43** is in a state of being positioned at the apex of the channel on/off spool chamber **23**; that is, in the initial state, and the lateral inlet **23***b* and the lateral outlet **23***c* are released via the opening area **43***c*, and the first communication channel **31** and the second communication channel **32** are in communication.

In addition, as a result of oil flowing to the upstream branch channel **13**, which is in communication with the channel on/off spool chamber **23** at the apex, and the oil pressure increasing, the channel on/off spool **43** slides against the elastic biasing force of the elastic member **47**, the first sliding part **431** reaches and closes the position of the lateral inlet **23***b* and the lateral outlet **23***c*, and blocks the first communication channel **31** and the second communication channel **32**.

When the channel on/off spool **43** slides by the pressure of the oil flowing through the upstream branch channel **13**, the first and second sliding parts **431**, **432** of the channel on/off spool **43** block the lateral inlet **23***b* and the lateral outlet **23***c* of the channel on/off spool chamber **23**, and block the communicating state of the first communication channel **31** and the second communication channel **32**. In addition, the flow of oil from the communication channel **3** to the channel cross-sectional area adjustment spool chamber **21** is stopped.

The channel cross-sectional area adjustment spool **41** is mounted on the channel cross-sectional area adjustment spool chamber **21** slidably in the axial direction and so as to cut across the main channel **11** in a substantially orthogonal state. The diameter of the first sliding part **411** (and the second sliding part **412**) of the channel cross-sectional area adjustment spool **41** is formed to be substantially equal to the inner diameter of the main channel **11**. In addition, as a result of the channel cross-sectional area adjustment spool **41** sliding in the axial direction, the protrusion of the constricted part **41***b* and the protrusion of the first sliding part **411** are increased/decreased in the main channel **11**, and the channel cross-sectional area of the main channel **11** is consequently contracted from a fully opened state.

The channel cross-sectional area adjustment spool **41** is normally subject to the elastic biasing force of the elastic member **45** so that the constricted part **41***b* cuts across within the main channel **11** and the channel cross-sectional area of the main channel **11** is fully opened to maximum. In addition, as a result of oil flowing into the channel cross-sectional area adjustment spool chamber **21**, the large diameter flange-shaped part **41***d* of the channel cross-sectional area adjustment spool **41** is pressed, and slides against the elastic biasing force of the elastic member **45**.

With the second hydraulic control valve A, in a low rotational range of the engine, the channel cross-sectional area adjustment spool **41** is in its initial state by the elastic member **45**, the constricted part **41***b* is in a fully open state in a state of cutting across the main channel **11**, and the entire amount of the oil passes through the opening area **41***c* around the constricted part **41***b* of the channel cross-sectional area adjustment spool **41** and flows from the upstream side to the downstream side (refer to FIG. **6**B).

In a low rotational range of the engine, the oil flowing through the main channel **11** may flow into the downstream branch channel **12** and the upstream branch channel **13**, but the channel on/off valve **42** and the channel on/off spool **43** will never engage in an on/off operation. Accordingly, there is no particular change in the hydraulic pressure, and the upper hydraulic pressure and the lower hydraulic pressure are substantially equal.

Subsequently, in a mid rotational range of the engine, the pressure of oil flowing from the main channel **11** to the downstream branch channel **12** will increase (refer to FIG. 7B). In addition, pursuant to the increase of pressure, the channel on/off valve **42** is pressed against the elastic biasing force of the elastically biasing elastic member **46**, and causes the channel on/off valve chamber **22** to slide. Consequently, the apex opening **22***a* and the lateral outlet **22***b* of the channel on/off valve chamber **22** are released, and the downstream branch channel **12** and the first communication channel **31** of the communication channel **3** are in communication.

Moreover, while the oil flowing through the main channel **11** also flows through the upstream branch channel **13**, the force from the hydraulic pressure on the upstream side in a mid rotational range is smaller than the elastic biasing force of the elastic member **47** that elastically biases the channel on/off spool **43**, and is maintained to be substantially immovable. In this state, the channel on/off spool chamber **43** is maintained in a substantial initial state, the constricted part **43***b* of the channel on/off spool **43** is positioned at the lateral inlet **23***b* and the lateral outlet **23***c* of the channel on/off spool chamber **23**, and the lateral inlet **23***b* and the lateral outlet **23***c* are of an open state.

Consequently, the downstream branch channel **12**, the first communication channel **31**, and the second communication channel **32** are in communication, and, through the downstream branch channel **12** and the communication channel **3** (first communication channel **31**, second communication channel **32**), oil flows from the apex inlet **21***a* of the channel cross-sectional area adjustment spool chamber **21** (refer to FIG. 7B). Moreover, in the foregoing case, the drain inlet **23***d* and the drain outlet **23***e* of the channel on/off spool chamber **23** are closed by the second sliding part **432** of the channel on/off spool **43** (refer to FIG. 7B).

Accordingly, with the channel cross-sectional area adjustment spool chamber **21**, oil will not flow out from the apex outlet **21***b*. Consequently, the channel cross-sectional area adjustment spool **41** slides against the elastic biasing force of the elastic member **45**. In addition, with the channel cross-sectional area adjustment spool **41**, the portion that cuts across the main channel **11** changes from the constricted part **41***b* to the first sliding part **411**, and the channel cross-sectional area of the main channel **11** is reduced (refer to FIG. 7B).

In other words, as a result of the channel cross-sectional area adjustment spool **41** sliding, the first sliding part **411** contracts the channel cross-sectional area of the main channel **11** and functions as an orifice. Accordingly, the flow rate and pressure of the oil flowing from the upstream side to the downstream side of the main channel **11** will decrease. However, the flow of oil is not completely stopped, and is only reduced, and a slight flow is maintained. Thus, as a result of the channel cross-sectional area of the main channel **11** decreasing, the hydraulic pressure will be lower in the downstream pressure (lower hydraulic pressure) of the control valve than the upstream pressure (equivalent to upper hydraulic pressure) of the control valve.

Subsequent, in a high rotational range of the engine, the pressure of oil on the upstream side of the main channel **11** will rise, and the pressure of oil flowing from the main channel **11** to the upstream branch channel **13** will also rise (refer to FIG. 8B). Consequently, the force from the pressure of oil flowing from the apex opening **23***a* of the channel on/off spool chamber **23** causes the channel on/off spool **43** to slide against the elastic biasing force of the elastic member **47** which elastically biases the channel on/off spool **43**.

In addition, the first sliding part **431** of the channel on/off spool **43** blocks the lateral inlet **23***b* and the lateral outlet **23***c* of the channel on/off spool chamber **23**, and the constricted part **43***b* simultaneously reaches the position of the drain inlet **23***d* and the drain outlet **23***e* and releases the drain inlet **23***d* and the drain outlet **23***e*.

Consequently, the channel cross-sectional area adjustment spool **41** is pressed by the elastic biasing force of the elastic member **45**, and the oil accumulated in the channel cross-sectional area adjustment spool chamber **21** flows from the apex outlet **21***b* through the drain channel **33**, flows through the drain inlet **23***d* and the drain outlet **23***e* of the channel on/off spool chamber **23**, and is discharged from the discharge channel **34** to the outside of the housing. The channel cross-sectional area adjustment spool **41** thereby smoothly returns to its initial position.

The first hydraulic control valve (2-stage relief valve) B is now explained. The first hydraulic control valve (2-stage relief valve) B is a device that operates only by hydraulic pressure, and does not include any electrically controlled structure. The 2-stage relief valve B is mainly configured from a valve housing **5** and a valve body **6**.

A valve passage **51** for sliding the valve body **6** is formed inside the valve housing **5**, and the valve body **6** slides through the valve passage **51**. A relief inflow part **52**, into which flows the oil discharged from the oil pump **9**, is formed at the end of valve housing **5** in an axial direction, and the relief inflow part **52** and the valve passage **51** are in communication (refer to FIG. **10**, FIG. **6**A and so on).

A stepped part is formed between the valve passage **51** and the relief inflow part **52**, and the stepped part becomes a relief inflow blocking surface **53**. The boundary of the relief inflow part **52** and the valve passage **51** is a so-called starting end **51***a* of the valve passage **51**, and, with the valve passage **51** as the reference position, a state where a valve head **62** of the valve body **6** comes into contact with the relief inflow blocking surface **53** is the initial state of the valve body **6**.

A first discharge part **54** and a second discharge part **55** are formed at respectively different positions, in the axial direction, at substantially the intermediate position of the valve housing **5** in the axial direction. The second discharge part **55** is formed more on the side of the relief inflow part **52** than the first discharge part **54** (refer to FIG. **10**, FIG. **6**A).

The first discharge part **54** is a through-hole which communicates the inside and outside of the valve housing **5**. The second discharge part **55** is formed at a position that is more on the side of the relief inflow part **52** than the first discharge part **54** in the passage direction of the valve passage **51**.

The valve body **6** is configured from an outer peripheral lateral part **61** and a valve head **62**, and, with the valve head **62**, a slope **62***b* is formed at the outer peripheral edge of a vertex **62***a*. The valve body **6** housed in the valve housing **5** is constantly elastically biased toward the relief inflow part **52** of the valve passage **51** with a spring **65** mounted on the valve passage **51**, and the valve head **62** of the valve body **6** comes into contact with the relief inflow blocking surface **53** of the valve passage **51**.

A substantial head-cut conical shape is formed with the vertex **62***a* and the slope **62***b*. A valve channel **63** is formed about the axis extending from the vertex **62***a* of the valve head **62** to the outer peripheral lateral part **61**. With regard to the valve channel **63**, a horizontal channel **63***c* is formed inside the valve body **6** along the axial direction from the valve head **62**, and a vertical channel **63***d*, which is orthogonal to the horizontal channel **63***c*, is formed around the horizontal channel **63***c* (refer to FIG. **6**A and so on).

In addition, the horizontal channel **63***c* is in communication with a head opening **63***b* formed on the valve head **62** and the vertical channel **63***d* is in communication with an outer peripheral lateral opening **63***a* of the outer peripheral lateral part **61**, and, with this kind of configuration, the head opening **63***b* and the outer peripheral lateral opening **63***a* are also in communication. The outer peripheral lateral opening **63***a* is formed in the outer peripheral lateral part **61** as an outer peripheral groove along the peripheral direction of the outer peripheral lateral part **61**.

The oil that is fed through the horizontal channel **63***c* and the vertical channel **63***d* flows out to the outer peripheral lateral opening **63***a*, which is formed as the outer peripheral groove, and the valve body **6** slides within the valve passage **51**, and the oil is fed to the first discharge part **54** in a state where the outer peripheral lateral opening **63***a* is in communication with the first discharge part **54**. With the spring **65**, one end thereof in the longitudinal direction is mounted on a spring support shaft **64** at the rear side of the valve body **6**, and the other end thereof is fixed by a holding member **56** mounted on the valve passage **51**. The outer peripheral lateral opening **63***a* of the valve body **6** is in communication with the valve channel **63** and the first discharge part **54** in a state of having reached the position of the first discharge part **54** formed in the valve housing **5**.

As described above, the second discharge part **55** is formed at a position that is more on the side of the relief inflow part **52** than the first discharge part **54**. In addition, in the initial state where the valve head **62** of the valve body **6** is in contact with the relief inflow blocking surface **53**, the second discharge part **55** is formed at a position that is more on the side of the relief inflow part **52** than the outer peripheral lateral opening **63***a* of the valve body **6**.

Accordingly, the outer peripheral lateral opening **63***a* of the valve body **6** is structured such that the outer peripheral lateral opening **63***a* communicates only with the first discharge part **54**, and does not communicate with the second discharge part **55**, as a result of the valve body **6** sliding within the valve passage **51**.

In addition, the valve body **6** is configured to slide by the hydraulic pressure of the oil that flows in from the relief inflow part **52** from the initial state, and, after the outer peripheral lateral opening **63***a* communicates with the first discharge part **54**, the valve head **62** of the valve body **6** passes through the second discharge part **55**. Moreover, oil is never discharged simultaneously from the first discharge part **54** and the second discharge part **55**.

With regard to the relief operation of the first hydraulic control valve B, in a low rotational range of the engine, both the first discharge part **54** and the second discharge part **55** are closed, and the oil is not relieved (refer to FIG. **6**A). Thus, the hydraulic pressure rises substantially proportionate to the engine speed.

In a mid rotational range of the engine, the first discharge part **54** and the outer peripheral lateral opening **63***a* are in communication, and the oil is relieved (refer to FIG. **7**A). Thus, the rise in hydraulic pressure relative to the engine speed becomes moderate. Moreover, in a mid to high rotational range (transition range) of the engine, both the first discharge part **54** and the second discharge part **55** are closed, and the oil is not relieved (refer to FIG. **7**C). Thus, the hydraulic pressure suddenly rises in the transition range.

In a high rotational range of the engine, the valve head **62** moves more toward the back side than the second discharge part **55**, and the oil is relieved from the second discharge part **55** (refer to FIG. **8**A). Thus, the rise in hydraulic pressure relative to the engine speed becomes moderate.

The operation of the engine lubrication control system of the present invention is now explained. Note that idling (also referred to as an idle rotation) is also included in the rotating state of the engine. In an idling range, the vehicle is stopped and a traction load is not applied to the engine, but in a low rotational range to a high rotational range, a load is applied to the engine since the vehicle is running. Moreover, as the basic motion, the second hydraulic control valve A controls the hydraulic pressure of the cam shaft supply path Sk**1** and the crank shaft supply path Sk**2** to be lower than the hydraulic pressure that is controlled by the first hydraulic control valve B across a range of a predetermined engine speed.

Foremost, in a low rotational range of the engine, both the first hydraulic control valve B and the second hydraulic control valve A are not operated, and the entire amount of the oil is fed to the cam shaft supply path S**1** and the crank shaft supply path S**2** (refer to FIG. **2**, FIG. **6**). In FIG. **2** to FIG. **4**, the arrow shows the flow of oil, and the thickness of the line in the arrow indicates the size of the flow rate.

Moreover, in a low rotational range of the engine, the configuration may also be such that the second hydraulic control valve A is operated from an engine speed that is lower than the minimum engine speed in a predetermined engine speed range. According to this kind of configuration, by constricting the crank shaft supply path Sk**2** and the cam shaft supply path Sk**1**, more oil will flow to the other oil branch supply paths Sk (variable valve timing mechanism supply path Sk**3**, oil jet supply path Sk**4**).

Thus, the hydraulic pressure of the variable valve timing mechanism supply path Sk**3**, oil jet supply path Sk**4** is controlled to be a higher pressure than the hydraulic pressure corresponding to the engine speed. Thus, the hydraulic pressure that is needed in a hydraulic transmission such as a variable valve timing mechanism can be secured from a lower rotation side, and the range of the engine speed in which the hydraulic transmission will operate can be expanded.

Subsequently, in a mid rotational range of the engine, the second hydraulic control valve A is operated (at a lower speed) prior to the first hydraulic control valve B (refer to FIG. **3**, FIG. **7**). Accordingly, in the flow of oil from upstream to downstream in the second hydraulic control valve A, the flow rate thereof will decrease, and the downstream pressure will become substantially constant without increasing. In addition, the supply of oil to the crank shaft supply path S**2** will decrease, and the increase of pressure is inhibited.

Meanwhile, with the first hydraulic control valve B, while the flow rate and pressure of the oil will decrease in a mid rotational range, since the second hydraulic control valve A is operating in advance, the flow of oil to the cam shaft supply path Sk**1** and the crank shaft supply path Sk**2** will decrease, and more oil will flow to the other oil branch supply paths Sk (variable valve timing mechanism supply path Sk**3**, oil jet supply path Sk**4**) (refer to FIG. **3**, FIG. **7**). Thus, it is possible to more quickly reach a hydraulic pressure of a level (for instance, 350 kPa) that is required for operating the variable valve timing mechanism.

In the engine lubrication control system, the control valve A starts its operation in a mid rotational range when the hydraulic pressure is, for example, 150 kPa. The first hydraulic control valve B starts its operation in a mid rotational range when the hydraulic pressure is, for example, 350 kPa. These are set to hydraulic pressures that are of at least a level in which the valve timing control (VTC) described later is operable with the foregoing hydraulic pressures.

Moreover, in a high rotational range of the engine, as a result of the control operation of the first hydraulic control valve (2-stage relief valve) B being added, the flow rate of the oil will increase (refer to FIG. **4**), and the hydraulic pressure will suddenly rise. As a result of configuring the setting so that the second hydraulic control valve A is switched to a high rotational mode at a value (for example, between 350 and 600 kPa) of the hydraulic pressure midway during the sudden rise of the hydraulic pressure caused by the first hydraulic control valve (2-stage relief valve) B, the downstream hydraulic pressure of the cam shaft supply path Sk**1** and the crank shaft supply path Sk**2** can also be caused to suddenly rise in conjunction with the sudden rise of the upstream hydraulic pressure in the oil branch supply path Sk of the first hydraulic control valve B.

This state is indicated in the graph shown in FIG. **5**. Accordingly, the hydraulic control of the second hydraulic control valve A can be performed in conjunction with the hydraulic control of the first hydraulic control valve B.

Moreover, as described above, the second hydraulic control valve A directly uses its hydraulic pressure on the upstream side and the downstream side of the installed position of the channel cross-sectional area adjustment spool **41** in the main channel **11** and controls the flow rate by contracting and expanding (restoring) the channel cross-sectional area of the main channel **11**. Thus, as the pressure of the oil that is flowing downstream and upstream of the main channel **11**, a predetermined hydraulic pressure value **1** and a predetermined hydraulic pressure value **2** which is greater than the predetermined hydraulic pressure value **1** are set as the pressure range.

In addition, in the main channel, when the hydraulic pressure that is more upstream than the channel cross-sectional area adjustment spool **41** becomes the predetermined hydraulic pressure value **2**, which is greater than the predetermined hydraulic pressure value **1**, the channel cross-sectional area adjustment spool **41** is restored and the channel cross-sectional area of the main channel **11** is maximized. Consequently, the operation of the channel cross-sectional area adjustment spool **41** becomes accurate and highly responsive, and the friction of the oil pump **9** can be decreased without impairing the lubrication of the crank shaft.

Specifically, in FIG. **5**, the predetermined hydraulic pressure value **1** is set to 150 kPa, and the predetermined hydraulic pressure value **2** is set to 600 kPa. The contraction and restoration (expansion) of the channel cross-sectional area of the main channel **11** are performed in the foregoing range.

What is claimed is:

1. An engine lubrication control system, comprising:

an engine;

an oil pump which is driven by the engine;

an oil circuit which extends downstream from the oil pump; and

a plurality of oil branch supply paths which branch from the oil circuit and supply oil to each part of the engine,

wherein a hydraulically-driven first hydraulic control valve which controls, in a stepwise manner, a discharge pressure of the oil pump relative to a speed of the engine is disposed on the oil circuit,

wherein a hydraulically-driven second hydraulic control valve is disposed on at least one of the plurality of oil branch supply paths,

wherein a downstream hydraulic pressure of the second hydraulic control valve is controlled to be lower than a downstream hydraulic pressure of the first hydraulic control valve of the oil circuit at least across a predetermined engine speed range,

wherein the second hydraulic control valve is set to be switched to a high rotational mode at a value of the hydraulic pressure midway during a rise in the hydraulic pressure caused by the first hydraulic control valve, so that the downstream hydraulic pressure of the second hydraulic control valve is caused to rise in conjunction with the rise in the downstream hydraulic pressure of the first hydraulic control valve,

wherein the second hydraulic control valve includes a channel cross-sectional area adjustment spool which changes a channel cross-sectional area of a main channel of the crank shaft supply path, and

wherein, when the hydraulic pressure that is more upstream than the channel cross-sectional area adjustment spool becomes a second predetermined hydraulic pressure value, which is greater than a first predetermined hydraulic pressure value, the channel cross-sectional area adjustment spool is restored and the channel cross-sectional area of the main channel is maximized.

2. The engine lubrication control system according to claim **1**, wherein the second hydraulic control valve is disposed on a crank shaft supply path or a cam shaft supply path among the plurality of oil branch supply paths.

3. The engine lubrication control system according to claim **2**, wherein the downstream hydraulic pressure of the first hydraulic control valve of the oil circuit is controlled to be substantially the same as the downstream hydraulic pressure of the second hydraulic control valve, at an engine speed that is higher than the predetermined engine speed range.

4. The engine lubrication control system according to claim **2**, wherein the engine speed when an operation the second hydraulic control valve is started is lower than the engine speed when an operation of the first hydraulic control valve is started.

5. The engine lubrication control system according to claim **2**,

wherein the channel cross-sectional area adjustment spool decreases the channel cross-sectional area of the main channel when a downstream hydraulic pressure of the channel cross-sectional area adjustment spool is greater than a first predetermined hydraulic pressure value, and the channel cross-sectional area adjustment spool is restored such that the channel cross-sectional area of the main channel is maximized when a hydraulic pressure that is more upstream than the channel cross-sectional area adjustment spool is a second predetermined hydraulic pressure value, which is greater than the first predetermined hydraulic pressure value.

6. The engine lubrication control system according to claim **2**, wherein the second hydraulic control valve controls a hydraulic pressure of the cam shaft supply path and the crank shaft supply path to be lower than the hydraulic pressure that is controlled by the first hydraulic control valve across a range of a predetermined engine speed.

7. The engine lubrication control system according to claim **1**, wherein the downstream hydraulic pressure of the first hydraulic control valve of the oil circuit is controlled to be substantially the same as the downstream hydraulic pressure of the second hydraulic control valve, at an engine speed that is higher than the predetermined engine speed range.

8. The engine lubrication control system according to claim **7**, wherein the engine speed when an operation the second hydraulic control valve is started is lower than the engine speed when an operation of the first hydraulic control valve is started.

9. The engine lubrication control system according to claim **7**,

wherein the channel cross-sectional area adjustment spool decreases the channel cross-sectional area of the main channel when a downstream hydraulic pressure of the channel cross-sectional area adjustment spool is greater than a first predetermined hydraulic pressure value, and the channel cross-sectional area adjustment spool is restored such that the channel cross-sectional area of the main channel is maximized when a hydraulic pressure that is more upstream than the channel cross-sectional area adjustment spool is a second predetermined hydraulic pressure value, which is greater than the first predetermined hydraulic pressure value.

10. The engine lubrication control system according to claim **1**, wherein the engine speed when an operation the second hydraulic control valve is started is lower than the engine speed when an operation of the first hydraulic control valve is started.

11. The engine lubrication control system according to claim **10**,

wherein the channel cross-sectional area adjustment spool decreases the channel cross-sectional area of the main channel when a downstream hydraulic pressure of the channel cross-sectional area adjustment spool is greater than a first predetermined hydraulic pressure value, and the channel cross-sectional area adjustment spool is restored such that the channel cross-sectional area of the main channel is maximized when a hydraulic pressure that is more upstream than the channel cross-sectional area adjustment spool is a second predetermined hydraulic pressure value, which is greater than the first predetermined hydraulic pressure value.

12. The engine lubrication control system according to claim **1**,

wherein the channel cross-sectional area adjustment spool decreases the channel cross-sectional area of the main channel when a downstream hydraulic pressure of the channel cross-sectional area adjustment spool is greater than a first predetermined hydraulic pressure value, and the channel cross-sectional area adjustment spool is restored such that the channel cross-sectional area of the main channel is maximized when a hydraulic pressure that is more upstream than the channel cross-sectional area adjustment spool is a second predetermined hydraulic pressure value, which is greater than the first predetermined hydraulic pressure value.

13. The engine lubrication control system according to claim **1**, wherein in a mid-rotational range of the engine, the second hydraulic control valve is operated at a lower speed prior to the first hydraulic control valve being operated.

14. The engine lubrication control system according to claim **1**, wherein in a low-rotational range of the engine, the second hydraulic control valve is operated from an engine speed that is lower than a minimum engine speed in a predetermined engine speed range.

15. The engine lubrication control system according to claim **1**, wherein hydraulic control of the second hydraulic control valve is performed in conjunction with hydraulic control of the first hydraulic control valve.

16. The engine lubrication control system according to claim **1**, wherein, in a low rotational range of the engine, both the first hydraulic control valve and the second hydraulic control valve are not operated.

17. The engine lubrication control system according to claim **1**, wherein the second hydraulic control valve is disposed on a crank shaft supply path or a cam shaft supply path among the plurality of oil branch supply paths,

wherein, in a low rotational range of the engine, both the first hydraulic control valve and the second hydraulic control valve are not operated such that an entire amount of oil is fed to the cam shaft supply path and the crank shaft supply path.

18. The engine lubrication control system according to claim **1**, wherein the first hydraulic control valve and the second hydraulic control valve are hydraulically driven and controlled in conjunction with each other.

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